

PEPC LRU: Ball Support Assembly

Terry Alger

May 14, 1999

U.S. Department of Energy



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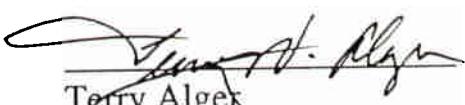
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MECHANICAL ENGINEERING SAFETY NOTE
MESN98-043-OA

May 14, 1999

PEPC LRU: Ball Support Assembly

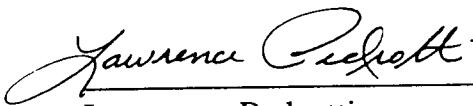
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Description:

The PEPC LRU upper ball support assembly consists of a ball and a pneumatic air cylinder/conical seat latching mechanism (see AAA97-109105) to be attached to the optics support frame, and a ball attached to the PEPC LRU (see AAA97-107592). Both components are designed to allow manual positioning in three axes. Upon insertion of the PEPC LRU into the structure, the upper pneumatic cylinder is actuated to latch the two assemblies together through the conical seat device to “grab” the lower ball to support the LRU weight. To be conservative, the design load for the assembly is 1500 pounds (the prototype PEPC LRU weight was measured to be near 1380 pounds).

Hazards:

Since the ball support assembly provides the only vertical support for the PEPC LRU, it represents a potential hazard to personnel and equipment should it fail. The PEPC LRU is mounted overhead so a failure would drop the LRU, causing equipment damage and potential personnel injury for anyone working beneath the equipment.

Design Calculations:

Table 1 lists the calculated maximum stresses, the required safety factor, and the calculated safety factors for the static 1500 pound design load for the different components of the ball support assembly. Table 2 presents the calculated maximum stresses and safety factors for the assembly components under seismic loading. According to the Mechanical Engineering Design Safety Standards, single bolt designs require a minimum safety factor of 6.0 based upon the material yield stress, and the minimum rare event safety factors required are 1.0 on yield stress or 1.25 on ultimate stress. The details of all calculations are presented in Appendix A.

CLASSIFICATION:		REVISIONS	
ZONE	KEY	DESCRIPTION	DRAWN BY
C3	QC1	ITEM 10 WAS HI20	CAB WF
B2	QC2	ADDED NIF NUMBERS	CAB WF
B3	QC3	BASELINE RELEASE UNDER CM	CAB WF

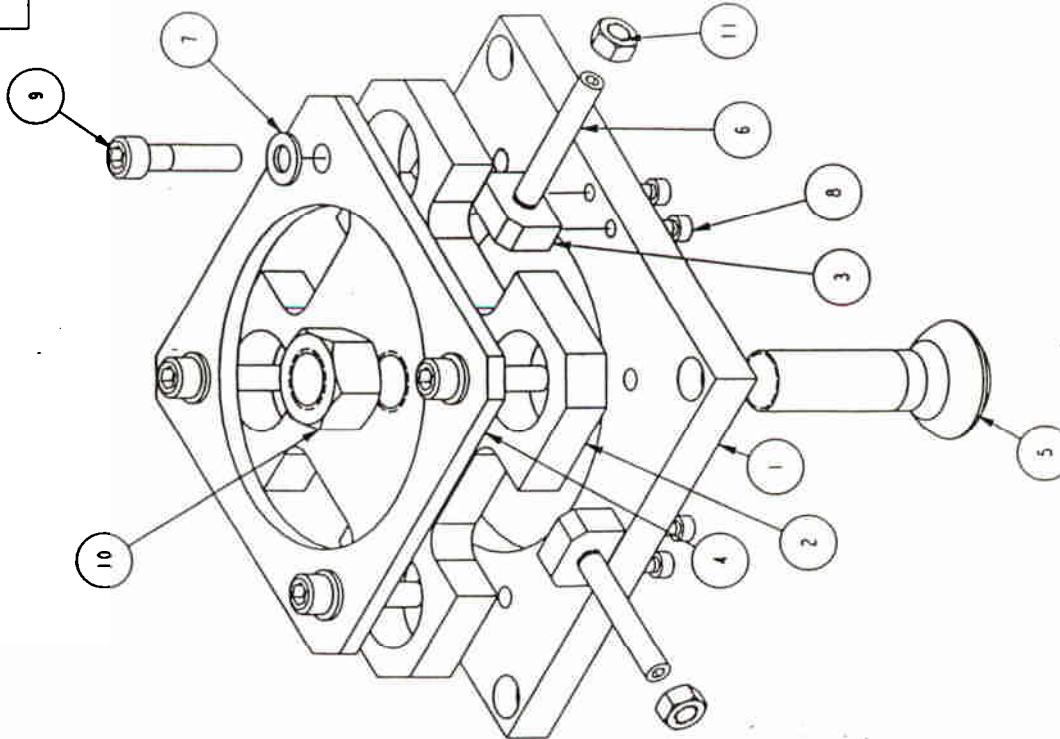
NOTES: UNLESS OTHERWISE SPECIFIED

102. APPLICABLE STANDARDS AND SPECIFICATIONS:
 ASME Y14.5M-1984, DIMENSIONING AND TOLERANCING
 ASME Y14.1, ABBREVIATIONS
 ASME Y14.36, SURFACE TEXTURE SYMBOLS
 NIF-0001270; AAN97-101357, NIF METRICATION POLICY

325. PRECISION CLEAN PARTS PER MEL98-009, PRECISION ULTRASONIC CLEANING FOR NIF COMPONENTS.
507. BAG THEN TAG WITH DRAWING NUMBER, TAB NUMBER AND SERIAL NUMBER IF APPLICABLE, AND REVISION LETTER IN 3 MM (.125 IN) HIGH CHARACTERS.

109. ESTIMATED WEIGHT IS 8.5 KG (18.7 LB).

4	N5310-10005	NUT, HEX, DIN934, M10	GR A2 SST	11
1	N5310-10009	NUT, HEX, DIN934, M24	GR A2 SST	10
4	N5305-10217	SCR, HEX SCH, DIN912 M10X45	GR A2 CL70	9
8	N5305-10199	SCR, HEX SCH, DIN912 M6X20	GR A2 CL70	8
4	N5310-10752	FLAT WASHER, REG, DIN 125 A 10MM	SST	7
4	N5340-12114	CARR LANE, SWIVEL SCREW CLAMP, TYPE B	SST	6
1	AAA97-107597	LRU BALL	A16 SST	5
1	AAA97-107596	RETAINER PLATE	1015 STL	4
4	AAA97-107595	TOMBSTONE	1015 STL	3
1	AAA97-107594-02	TRANSLATION PLATE, TAB-02	1020 STL	2
1	AAA97-107593	BASE PLATE	1020 STL	1
917 #000	IDENTIFYING NO.	AMMUNITION OR DELIVERY PARTS LIST	ITEM SPECIFICATION	



CLASSIFICATION: UNCLASSIFIED

DATE: 04/04/03
 DRAWING: AAA97-107592
 MODEL: AAA97-107592
 SHEET: 1 OF 1
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THIRD ANGLE PROJECTION				UNCLASSIFIED	WBS 1.3.3 NIF - The National Ignition Facility
		APPROVING	NAME: G. ZULETA	LAWRENCE LIVERMORE NATIONAL LABORATORY	UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA
		INTEGRATOR	NAME: M. FUNKHOUSER	PEPC LRU ASSEMBLY	
		TECHNICAL	NAME: M. FUNKHOUSER	LRU FRAME ASSEMBLY	
		DETAILED	NAME: M. FUNKHOUSER	KINEMATIC BALL ASSEMBLY	
		TEST	NAME: C. 14067	DOC. NO. AAA97-107592	-0C
		ASSEMBLY	DATE: 04/04/03	CLASS: NONE	SECTION: 1 OF 1
		MANUFACTURE	TIME: 11:00 AM	REVISION: 0	
		TEST ASSIST	SCALE: 1:1	DATE: 04/04/03	
		APP. SIGNATURE	SCALE: 1:1	CLASS: NONE	

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DATE: 04/04/03
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 APPROVAL: _____
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 DRAWING: AAA97-107592
 MODEL: AAA97-107592
 SHEET: 1 OF 1
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 DATE: 04/04/03

Table 1: Static 1500 pound Load Ball Support Assembly Component Part Stress and Safety Factor Summary

Ball Support Component	Calculated Stress (ksi)	Required Safety Factor	Calculated Safety Factor	Requirements
upper ball shank (see Appendix A, Section 3.0)	16.1 ✓	6.0 (based upon yield stress)	10.6 ✓ (based upon yield stress)	certified 17-4PH stainless steel in H900 condition
lower ball shank (see Appendix A, Section 4.0)	<16.1 ✓	6.0 (based upon yield stress)	>10.6 (based upon yield stress)	certified 17-4PH stainless steel in H900 condition
upper translation plate (see Appendix A, Section 5.0)	6.3 ✓	4.0 (based upon ultimate stress)	9.0 (based upon ultimate stress)	annealed 1020 steel or equivalent
lower translation plate (see Appendix A, Section 6.0)	<6.3 ✓	4.0 (based upon ultimate stress)	>9.0 (based upon ultimate stress)	annealed 1020 steel or equivalent
tie rod bending (see Appendix A, Sections 7.0 and 15.5)	6.3 ✓	none for safety, functional only	14.4 ✓ (based upon ultimate stress)	the tie rod diameter must be a minimum diameter of 0.5 inch and the material must be cold drawn 1045 steel or equivalent
tie rod end screws (see Appendix A, Sections 7.0 and 15.5)	10.9	none for safety, functional only	16.4 (based upon ultimate stress)	the attachment screws must be Textron CAMCAR steel allen head screws with a torque of 20-25 inch pounds

tie rod end screw/plate threads (see Appendix A, Sections 7.0 and 15.5)	3.2 ✓	none for safety, functional only	7.3 (based upon yield stress)	required 6061-T6 Al and a minimum of 0.25 inch thread engagement with a torque of 20-25 inch pounds
pneumatic cylinder rod bending (see Appendix A, Section 8.0)	8.5 ✓	none for safety, functional only	9.4 (based upon ultimate stress)	requires 300 SS or equivalent
pneumatic cylinder rod attachment screw (see Appendix A, Section 8.0)	8.6 ✓	none for safety, functional only	9.2 (based upon ultimate stress)	requires 300 SS or equivalent
pneumatic cylinder rod/plate threads (see Appendix A, Section 8.0)	4.4 ✓	none for safety, functional only	5.6 (based upon yield stress)	requires a minimum thread engagement of 0.375 inch in 1020 steel or better with a torque of 20-25 inch pounds
collar (see Appendix A, Section 9.0)	2.3 ✓	6.0 (based upon yield stress)	18.9 (based upon yield stress)	annealed 1020 steel or equivalent—it is critical that the collar be properly engaged during the maintenance cycle, and therefore, there should be an operator feedback device indicating its position
clamshell internal (see Appendix A, Section 10.0)	6.2	6.0 (based upon yield stress)	11.5 (based upon yield stress)	requires certified 7075-T6/T651 Al
base plate mounting	0.5 ✓	4.0 (based upon	43.0 (based	300 SS or better with a torque of 10

screws (see Appendix A, Section 12.0)		ultimate stress)	upon yield stress)	foot-pounds
clamshell at ball contact line (see Appendix A, Section 14.0)	~44.4	compressive stress, not a safety issue	1.6 (based upon yield stress)	certified 7075-T6/T651 Al

Table 2: Seismic Load Ball Support Assembly Component Part Stress and Safety Factor Summary

Ball Support Component	Calculated Stress (ksi)	Required Safety Factor	Calculated Safety Factor
upper ball shank (see Appendix A, Section 15.1)	86.0	1.25 (based upon ultimate stress)	2.2 (based upon ultimate stress)
lower ball shank (see Appendix A, Section 15.2)	<86.0	1.25 (based upon ultimate stress)	>2.2 (based upon ultimate stress)
upper translation plate (see Appendix A, Section 15.3)	15.2	1.25 (based upon ultimate stress)	3.7 (based upon ultimate stress)
lower translation plate (see Appendix A, Section 15.4)	<15.2	1.25 (based upon ultimate stress)	>3.7 (based upon ultimate stress)
tie rod bending (see Appendix A, Section 15.5)	33.8	none for safety, functional only	2.7 (based upon ultimate stress)

tie rod end screws (see Appendix A, Section 15.5)	58.5	none for safety, functional only	3.1 (based upon ultimate stress)
tie rod end screws/plate threads (see Appendix A, Section 15.5)	16.9	none for safety, functional only	1.4 (based upon yield stress)
pneumatic cylinder rod bending (see Appendix A, Section 15.6)	51.9	none for safety, functional only	1.5 (based upon ultimate stress) 0.7 (based upon yield stress)
pneumatic cylinder rod attachment screw (see Appendix A, Section 15.6)	70.4	none for safety, functional only	1.1 (based upon ultimate stress) 0.5 (based upon yield stress)
pneumatic cylinder rod attachment screw/plate thread stress (see Appendix A, Section 15.6)	27.0	none for safety, functional only	0.9 (based upon yield stress)
collar (see Appendix A, Section 15.7)	5.0	1.25 (based upon ultimate stress)	11.3 (based upon ultimate stress)
clamshell internal (see Appendix A, Section 15.8)	13.8	1.25 (based upon ultimate stress)	5.9 (based upon ultimate stress)
base plate mounting screws (see calculation Section 15.10)	2.9	1.00 (based upon yield stress)	7.0 (based upon yield stress)
clamshell at ball	-66.0	compressive	1.2 (based upon

contact line (see Appendix A, Section 15.12)		stress, not a safety issue	ultimate stress) 1.1 (based upon yield stress)
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Summary:

It can be seen from the summary tables that all safety related system components have safety factors that are greater than those required by the LLNL Mechanical Engineering Design Safety Standards. Therefore, the PEPC LRU upper ball support assembly should be approved for use in a manned area up to the 1500 pound design load.

Testing and Labeling:

The completed assembly shall be tested to 1.5 times the design load, i.e., simultaneously applied 2250 pound vertical and 236 pound horizontal loads located at the lower ball position. The assembly shall be labeled, at a minimum, as follows:

LLNL LIMIT TESTED	
DWG. NO.:	AAA97-109105 and AAA97-107592
LOAD LIMIT :	1500 pounds
REFERENCE:	MESN98-043-OA
TESTED BY:	
WITNESSED BY:	
DATE:	

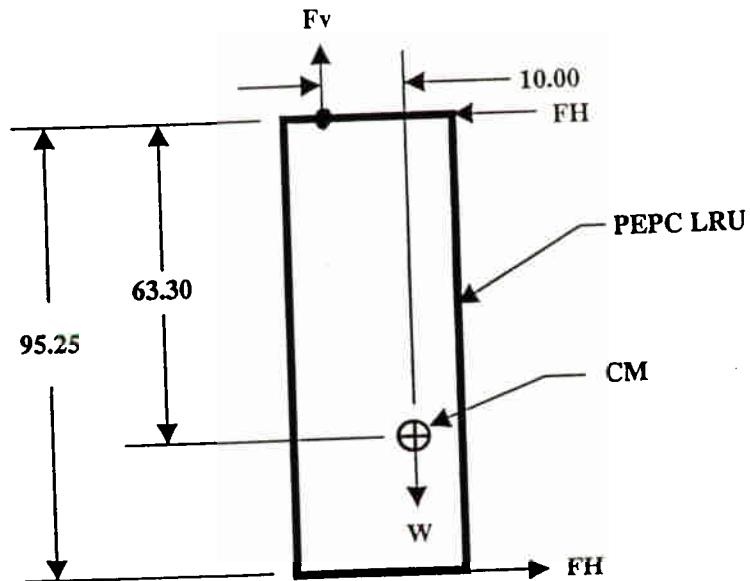
1.0 General Information:

Total System Weight: 1500 pounds (the measured prototype assembly weight was 1382 pounds)

Assembly Drawings: Ball Grabber Assembly: AAA97-019105
Kinematic Ball Assembly: AAA97-107592

2.0 PEPC LRU Load Determination:

The general PEPC LRU force schematic is



Summing vertical forces yields

$$F_v = W = 1500\#$$

Summing moments about the CM yields

$$FH(95.25 - 63.3) + FH(63.3) - F_v(10) = 0$$

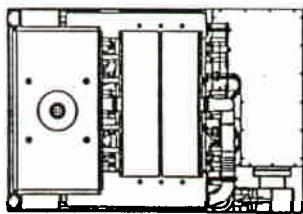
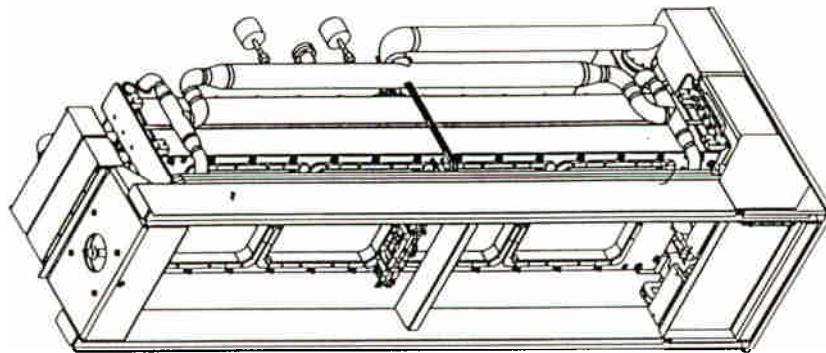
or

$$FH = 157\#$$

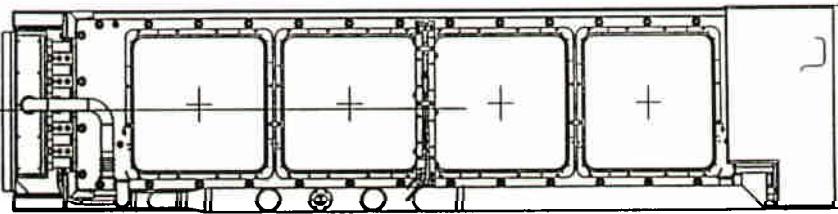
which is the horizontal force reaction at the upper kinematic ball assembly.

4/28/98
part

Y
Z

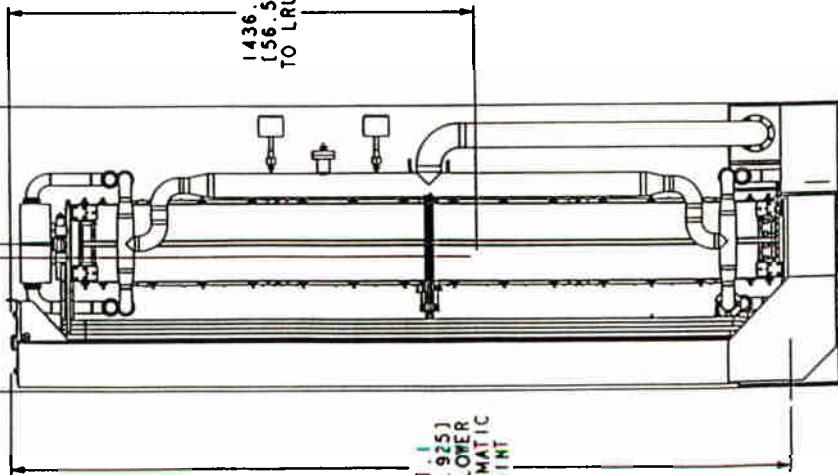


271.1
[10.673]
TO LRU CG



5.12
[20.21]
TO LRU CG

467.3
[18.3916]
432.7
[17.0354]



1436.84
[56.568]
TO LRU CG

2411.
[94.925]
TO LOWER
KINEMATIC
POINT

900±2
[35.433±.078]
618±2
[24.331±.078]

3.0 Stress in Upper Ball Shank (see AAA97-109110):

The upper ball threads (M24-3.0) into a plate with a length near 3.85 in. to the application of the side load FH. The ball shank has been machined flat on opposite sides. The axial load stress becomes

$$F_v = 1500 \quad \text{pounds, vertical load}$$

$$D = 0.780 \quad \text{inches, minimum shank diameter at top (i.e., conservative calculation)}$$

$$\sigma_a := \frac{F_v}{\frac{\pi}{4} \cdot D^2}$$

$$\sigma_a = 3.139 \cdot 10^3 \quad \text{psi, axial load stress at top (neglects pin cutout reduction in area since max. cutout nonexistent at max. stress location)}$$

The bending stress due to the side loading becomes

$$F_h = 157 \quad \text{pounds, horizontal load}$$

$$L = 3.85 \quad \text{inches, lever arm}$$

$$I = \frac{\pi}{4} \cdot \left(\frac{D}{2}\right)^4$$

$$I = 0.018 \quad \text{in.}^4, \text{ moment of inertia}$$

$$\sigma_b := \frac{32 \cdot F_h \cdot L}{\pi \cdot D^3}$$

$$\sigma_b = 1.297 \cdot 10^4 \quad \text{psi, bending stress}$$

Therefore, the total maximum axial tensile stress is

$$\sigma_m := \sigma_a + \sigma_b$$

$$\sigma_m = 1.611 \cdot 10^4 \quad \text{psi, total superposed stress}$$

The safety factors based become

$$\begin{aligned} \text{UTS} &:= 193000 \quad \text{psi, ultimate tensile stress of 17-4PH SS in} \\ &\quad \text{H900 condition (see pg. 3.14-5, in "Damage Tolerant Design Handbook", 1983)} \\ \text{YS} &:= 170000 \quad \text{psi, yield stress of 17-4PH SS in H900 condition} \end{aligned}$$

$$\text{SFu} := \frac{\text{UTS}}{\sigma_m} \qquad \text{SFy} := \frac{\text{YS}}{\sigma_m}$$

$$\text{SFu} = 11.978 \quad \text{SFy} = 10.55$$

Upper Ball Shank Fracture Critical Calculations:

$$\sigma_m = 1.611 \cdot 10^4 \quad \text{psi, tensile stress}$$

$$\begin{aligned} K_{1c} &:= 84 \quad \text{ksi in}^2, \text{ fracture toughness (see pg. 3.14-2, in "Damage Tolerant Design Handbook", 1983)} \\ a_c &:= \left(\frac{1}{\pi} \right) \cdot \left(\frac{K_{1c} \cdot 1000}{1.12 \cdot \sigma_m} \right)^2 \end{aligned}$$

$$a_c = 6.896 \quad \text{inches, critical flaw size}$$

$$\frac{\sigma_m}{Y_S} = 0.095 \quad \text{stress ratio}$$

For a Stress Ratio = 0.08 and initial flaw size of 0.125 inches (see Figure 3.14.3.1, pg. 3.14-5, in "Damage Tolerant Design Handbook", 1983):

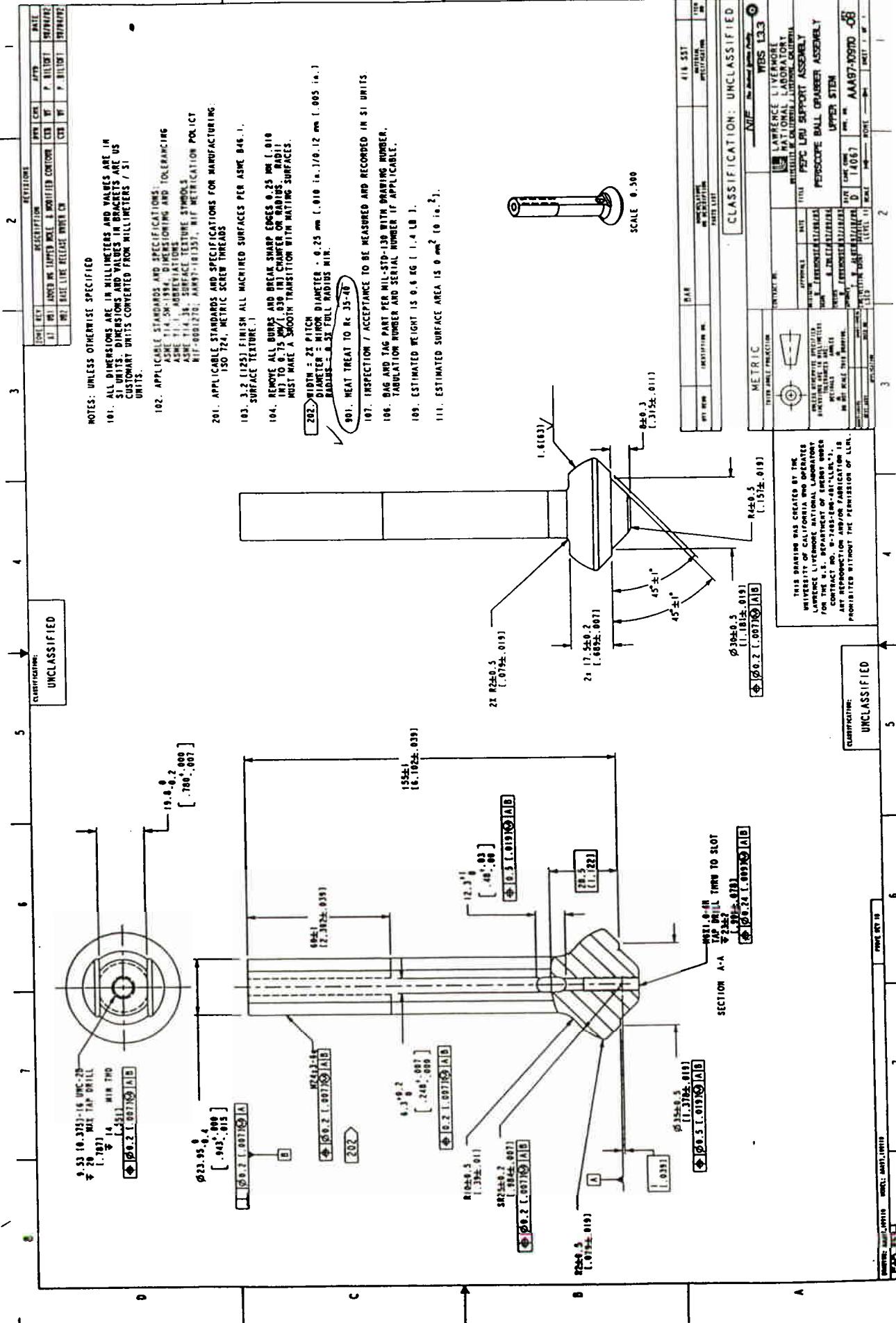
$$A := 2.606 \cdot 10^{-10} \quad (6.8 \times 10^{-10})$$

$$n := 3.192$$

$$a_0 := 0.125 \quad \text{inches, assumed initial crack size}$$

$$N_{cycles} = \left(\frac{2}{2-n} \right) \left[\frac{1}{A \cdot \left(1.12 \cdot \frac{\sigma_m}{1000} \cdot \sqrt{\pi} \right)^n} \right] \cdot \left[ac \left(\frac{2-n}{2} \right) - ao \left(\frac{2-n}{2} \right) \right]$$

$N_{cycles} = 3.173 \cdot 10^5$ ↗ number of maintenance loading cycles required to grow an assumed initial 0.125 inch flaw to critical flaw size



4.0 Stress in Lower Ball Shank (see AAA97-107597):

The lower ball shank has a smaller load lever arm. Therefore, the tensile stress will be less than that of the upper ball shank. Because of this, the stress will not be determined.

CLASSIFICATION: UNCLASSIFIED

ZONE	REV	DESCRIPTION	DRAWN	CHECKED	APPROVED	DATE
C4	0C1	95.0 DIM WAS .95 & .60 WAS .55	CB	WF	P. BILLOFF	8/07/30
C3	0C1	M2413 WAS M2012.5 & MODIFIED COMTOR	CB	WF	P. BILLOFF	8/07/30
	0C2	BASE LINE RELEASE UNDER CM	CB	WF	P. BILLOFF	8/07/30

NOTES: UNLESS OTHERWISE SPECIFIED

101. ALL DIMENSIONS ARE IN MILLIMETERS AND VALUES ARE IN SI UNITS. DIMENSIONS AND VALUES IN BRACKETS ARE US CUSTOMARY UNITS CONVERTED FROM MILLIMETERS / SI UNITS.

102. APPLICABLE STANDARDS AND SPECIFICATIONS:
ASME Y14.5M-1994, DIMENSIONING AND TOLERANCING
ASME Y1.1, ABBREVIATIONS
ASME Y14.36, SURFACE TEXTURE SYMBOLS
NIF-0001210; AAN97-101357, NIF METRICATION POLICY

103. APPLICABLE STANDARDS AND SPECIFICATIONS FOR MANUFACTURING:
ISO 724, METRIC SCREW THREADS
ISO 1251 FINISH ALL MACHINED SURFACES PER ASME B4.6 - 1985, SURFACE TEXTURE.

104. REMOVE ALL BURRS AND BREAK SHARP EDGES 0.25 MM (.010 IN) TO 0.15 MM (.030 IN) CHAMFER OR RADIUS. RADII MUST MAKE A SMOOTH TRANSITION WITH MATING SURFACES.

105. HEAT TREAT TO RC 35-40

106. BAG AND TAG PART PER MIL-STD-130 WITH DRAWING NUMBER, TABULATION NUMBER AND SERIAL NUMBER IF APPLICABLE, AND REVISION LETTER.

107. INSPECTION / ACCEPTANCE TO BE MEASURED AND RECORDED IN SI UNITS.

108. ESTIMATED WEIGHT IS 0.5 kg (.10 lb).

109. ESTIMATED SURFACE AREA IS 10592 mm² (16 in.²).

CLASSIFICATION: UNCLASSIFIED

METRIC	BAR	BAR	416 SST	MATERIAL SPECIFICATION NO.
OTI RECD	IDENTIFY NO.	MANUFACTURE OR DESIGNER PARTS LIST		

CLASSIFICATION: UNCLASSIFIED

WBS 1.3.3

LAWRENCE LIVERMORE NATIONAL LABORATORY
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NATIONAL LABORATORY
KINEMATIC BALL ASSEMBLY
LRU BALL

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5.0 Stress in Upper Translation Plate (see AAA97-107594):

The stress in the upper translation plate can be approximated by superposing the stresses in a circular, fixed edge plate with both central pressure and central moment loading conditions. The central area has a radius $b=0.408$ in. (ball shank thread root radius) and the plate outer radius is $a=2.657$ in. (clamp plate hole radius). The plate stress is then (see 4th Ed. Roark , #7, pg. 218 for the pressure load case and 5th Ed. Roark, #21, pg. 435 for the moment load case)

$$W_v := 1500 \text{ pounds, applied vertical load} \quad a := 2.657$$

$$W_h := 157 \text{ pounds, applied horizontal load} \quad b := 0.408$$

$$L := 3.85 \text{ inches, side load moment lever arm} \quad \frac{b}{a} = 0.154$$

$$M := W_h \cdot L \text{ in. pounds, side load moment} \quad M = 604.45$$

$$\nu := 0.3 \quad \text{Poisson's ratio}$$

$$m := \frac{1}{\nu} \quad m = 3.333$$

$$t := 0.75 \text{ inches, plate thickness}$$

$$\sigma_p := \frac{3 \cdot W_v \cdot (m + 1)}{2 \cdot \pi \cdot m \cdot t^2} \left[\ln\left(\frac{a}{b}\right) + \left(\frac{b^2}{4 \cdot a^2} \right) \right]$$

$$\sigma_p = 3.111 \cdot 10^3 \text{ psi, maximum plate tensile stress due to center pressure load}$$

$$\gamma := 7.9 \quad \text{per Roark 5th Edition for } b/a = 0.15$$

$$\sigma_m := \frac{\gamma \cdot M}{a \cdot t^2}$$

$$\sigma_m = 3.195 \cdot 10^3 \text{ psi, maximum plate tensile stress due to center moment}$$

Therefore, the combined stress results with

$$\sigma := \sigma_p + \sigma_m$$

$$\sigma = 6.306 \cdot 10^3 \checkmark \quad \text{psi, maximum plate tensile stress due to combined loads}$$

This results with the safety factors as

$$UTS := 57000 \quad \text{psi, ultimate tensile stress of annealed 1020 steel (LLNL M.E. DSS)}$$

$$YS := 43000 \quad \text{psi, yield stress of 1020 steel}$$

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 9.039 \quad SF_y = 6.819 \checkmark$$

6.0 Stress in Lower Translation Plate (see AAA97-107594):

The stress in the lower translation plate will be less than that of the upper translation plate, and therefore will not be determined.

7.0 Stress in Tie Rods (see AAA97-109113)*:

Because of the side load applied to the kinematic balls, there is a couple applied to the tie rods through the collar (see AAA97-109106). Under load, the kinematic balls are separated by 1.3 inches, resulting with the couple as

$$W_h = 157 \quad \text{pounds, applied horizontal load}$$

$$M := 1.3 \cdot W_h$$

$$M = 204.1 \quad \text{in. pounds, value of couple}$$

There are three tie rods, 0.375 in. in diameter, supporting the couple and effectively cantilevered. Therefore, the max. tensile stress in the rods is

$$I := \frac{\pi}{4} \cdot \left(\frac{0.375}{2} \right)^4$$

$$I = 9.707 \cdot 10^{-4} \text{ in.}^4, \text{ moment of inertia of a rod}$$

$$N := 3 \quad \text{number of rods}$$

$$\sigma := \frac{M \cdot \left(\frac{0.375}{2} \right)}{N \cdot I}$$

$$\sigma = 1.314 \cdot 10^4 \quad \text{psi, max. rod bending stress}$$

The safety factors are

$$UTS := 80000 \quad \text{psi, ultimate tensile stress of 300 SS (M. E. Handbook, 7th Ed., page 6-43)}$$

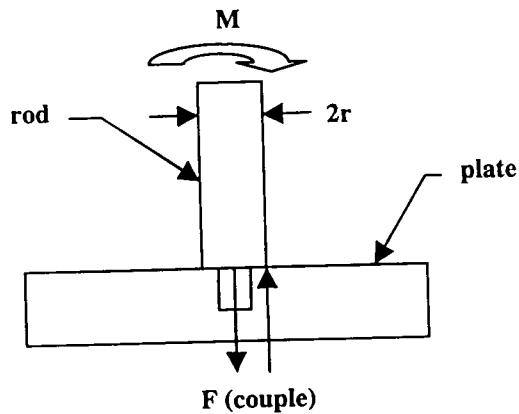
$$YS := 35000 \quad \text{psi, yield stress of 300 SS}$$

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 6.088 \quad SF_y = 2.663$$

It should be noted that the failure of these rods does not represent a safety issue; only an inconvenience in repair. Failure of the rods will not release the load, and in fact, failure of the rods will make it so the load cannot be released.

The stress in the threaded tie rod ends can be approximated according to the arrangement



where

$$M = 204.1 \quad \text{in. pounds, moment on rod}$$

$$r := \frac{0.375}{2} \quad \text{in., rod radius}$$

$$F := \frac{M}{N \cdot r}$$

$$F = 362.844 \quad \text{pounds, load per rod end screw}$$

The tensile stress in the screw (0.25-20, root dia. near 0.19 in.) is

$$\sigma := \frac{F}{\frac{\pi}{4} \cdot (0.19)^2}$$

$$\sigma = 1.28 \cdot 10^4 \quad \checkmark \quad \text{psi, rod end screw tensile stress}$$

The safety factors are

$$SF_u := \frac{UTS}{\sigma} \qquad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 6.251 \quad SF_y = 2.735$$

*Note: See Section 15.5 for calculations concerning a required increase in the rod diameter from 0.375 in. to 0.50 in., and a required change in the material from 300 SS to cold drawn 1045 steel. Also note that CAMCAR steel allen head screws are also required. These changes are necessary to meet the seismic requirements presented later in Section 15.

8.0 Stress in Pneumatic Cylinder Rod and Rod Screw:

The same couple determined in Section 7.0 must be resisted by the pneumatic cylinder rod and rod end screw. The pneumatic cylinder rod is 0.625 in. diameter 300 SS with a 0.375-16 threaded end (screw root dia. is 0.31 in.). The rod is threaded into the ball shank/support plate assembly and must resist the couple created by the horizontal load. Similar to section 7.0, we have

$$M := 204.1 \text{ in. pounds, value of couple}$$

$$r := \frac{0.625}{2}$$

$$I := \frac{\pi}{4} \cdot r^4$$

$$I = 7.49 \cdot 10^{-3} \text{ in.}^4, \text{ moment of inertia of the rod}$$

$$\sigma := \frac{M \cdot r}{I}$$

$$\sigma = 8.515 \cdot 10^3 \text{ psi, max. rod bending stress}$$

The safety factors are

$$UTS := 80000 \text{ psi, ultimate tensile stress of 300 SS (M.E. Handbook, 7th Ed., page 6-43)}$$

$$YS := 35000 \text{ psi, yield stress of 300 SS}$$

$$SF_u := \frac{UTS}{\sigma}$$

$$SF_y := \frac{YS}{\sigma}$$

$$SF_u = 9.395$$

$$SF_y = 4.11$$

Again, it should be noted that failure of this rod does not represent a safety issue---only an inconvenience in repair. Failure of the rod will not release the load, but could require a repair of the hardware prior to further use.

The stress in the end screw is

$$M = 204.1 \quad \text{in. pounds, moment on rod}$$

$$F = \frac{M}{r}$$
$$F = 653.12 \quad \text{pounds, end screw load}$$

$$\sigma := \frac{F}{\frac{\pi}{4} \cdot (0.31)^2}$$

$$\sigma = 8.653 \cdot 10^3 \quad \text{psi, rod end screw tensile stress}$$

The safety factors are

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 9.245 \quad SF_y = 4.045$$

The end screw/plate thread stress is

$$L := 0.375 \quad \text{in., minimum thread engagement length}$$

$$\tau_{st} := \frac{F}{\left(\frac{\pi \cdot 0.25 \cdot L}{2} \right)}$$
$$\tau_{st} = 4.435 \cdot 10^3 \quad \text{psi, end screw thread shear stress}$$

which results with the safety factor

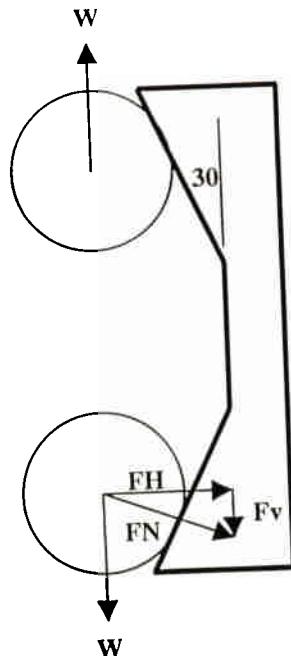
$$YS := 43000 \quad \text{psi, yield stress of annealed 1020 steel (LLNL M.E. DSS)}$$

$$SF := \frac{YS}{\sqrt{3 \cdot \tau_{st}}}$$

$$SF = 5.598$$

9.0 Stress in Collar (see AAA97-109106):

The stress in the collar can be approximated by assuming that all the horizontal force created by the ball loading to the clamshells is supported as an internal pressure by the collar. The force transferred from the clamshells to the collar can be determined from the geometry



where the horizontal force is

$$W_v := 1500 \quad \text{pounds, vertical load}$$

$$F_h := \frac{W_v}{\tan\left(30 \cdot \frac{\pi}{180}\right)}$$

$$F_h = 2.598 \cdot 10^3 \quad \text{pounds, horizontal force at each ball}$$

Therefore, the total horizontal force supported by the collar is

$$F_{ht} := 2 \cdot F_h$$

$$F_{ht} = 5.196 \cdot 10^3 \quad \text{pounds, total horizontal force}$$

The effective pressure is then

$$H := 2.0 \text{ in., effective clamshell "load" height}$$

$$ID = 2.76 \text{ in., collar inside dia.}$$

$$OD = 3.15 \text{ in., collar outside dia.}$$

$$P := \frac{F_{ht}}{\pi \cdot ID \cdot H}$$

$$P = 299.635 \text{ psi, effective collar internal pressure}$$

The maximum collar hoop stress is then (assuming a thick walled pressure vessel)

$$\sigma := P \cdot \frac{(OD^2 + ID^2)}{(OD^2 - ID^2)}$$

$$\sigma = 2.28 \cdot 10^3$$

and safety factors are

$$UTS := 57000 \text{ psi, ultimate tensile stress of annealed 1020 steel (LLNL M.E. DSS)}$$

$$YS := 43000 \text{ psi, yield stress of 1020 steel}$$

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 24.998 \quad SF_y = 18.858$$

10.0 Stress in Clamshell (see AAA97-109111):

The stress in the clamshell can be determined by treating it as a beam with an applied axial force (i.e., the 1500 # vertical load) that creates a tensile stress due to both an axial load and an end couple (see schematic in Section 9.0). This model will yield a conservative prediction of the tensile stress since it assumes that none of the end couple will be resisted by the cylindrical support collar.

The uniform tensile stress caused by the axial load is

$$W_v := 1500 \text{ pounds, vertical load}$$

$$ID := 2.166 \text{ in., inside diameter} \quad R_i := \frac{ID}{2} \quad R_i = 1.083$$

$$OD := 2.756 \text{ in., outside diameter} \quad R_o := \frac{OD}{2} \quad R_o = 1.378$$

$$c := 0.394 \text{ in., cutout width}$$

$$\theta := \left(\operatorname{atan} \left(\frac{c}{\frac{OD + ID}{4}} \right) \right) \cdot \left(\frac{180}{\pi} \right) \text{ degrees of cutout at ave. radius}$$

$$\theta = 17.755$$

$$CF := \frac{4 \cdot \theta}{360} \text{ cutout fraction} \quad CF = 0.197$$

$$AF := 1 - CF \text{ area fraction} \quad AF = 0.803$$

$$\sigma_a := \frac{W_v}{\frac{\pi}{4} \cdot (OD^2 - ID^2) \cdot AF}$$

$$\sigma_a = 819.297 \text{ psi, tensile stress from axial load}$$

The maximum tensile stress due to the end couple is

$$\theta_{max} := \left(\operatorname{atan} \left(\frac{c}{\frac{ID}{2}} \right) \right) \cdot \left(\frac{180}{\pi} \right) \text{ degrees, cutout angle at minimum radius}$$

$$\theta_{max} = 19.992$$

$$\theta_{max} := \theta_{max} \cdot \frac{\pi}{180} \text{ radians} \quad Y_o := c \text{ in., cutout width}$$

$$A := \left(\int_{Ri}^{Ro} r dr \right) \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$A = 0.887$ in², approx. area of minimum thickness section

$$YDA := \left[\int_{\theta_{max}}^{\pi - \theta_{max}} \sin(\theta) d\theta \right] \cdot \left(\int_{Ri}^{Ro} r^2 dr \right) - Yo \cdot \left(\int_{Ri}^{Ro} r dr \right) \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$YDA = 0.494$ in³, integral of $y^2 dA$

$$Ycm := \frac{YDA}{A}$$

$Ycm = 0.557$ in., distance to center of mass (area) from the cutout

$Y := Ycm + Yo$ $Y = 0.951$ in., distance to center of mass from clamshell center

$$Int1 := Y^2 \cdot \left(\int_{Ri}^{Ro} r dr \right) \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$$Int2 := 2 \cdot Y \cdot \left(\int_{Ri}^{Ro} r^2 dr \right) \left[\int_{\theta_{max}}^{\pi - \theta_{max}} \sin(\theta) d\theta \right]$$

$$Int3 := \left(\int_{Ri}^{Ro} r^3 dr \right) \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (\sin(\theta))^2 d\theta \right]$$

$Ixx := Int1 - Int2 + Int3$

$Ixx = 0.058$ in.^4, beam bending moment of inertia, i.e., integral of $y^2 dA$

$$L := \frac{0.295}{2} + \left(1.083 - \frac{1.969}{2} \cdot \cos\left(30 \cdot \frac{\pi}{180}\right) \right)$$

$L = 0.378$ in., couple force lever arm to clamshell wall centerline

$$M := Wv \cdot L$$

$M = 566.847$ in. pounds, applied end couple

$$\sigma_m := \frac{M \cdot Y_{cm}}{I_{xx}}$$

$\sigma_m = 5.421 \cdot 10^3$ psi, max. tensile stress due to end couple

Therefore, the total max. stress is the superposition of the above two stresses, i.e.,

$$\sigma_t := \sigma_a + \sigma_m$$

$\sigma_t = 6.241 \cdot 10^3$ psi, total superposed tensile stress

The safety factors are

$UTS := 82000$ ✓ psi, ultimate tensile stress of 7075-T6 Al (M. E. Handbook, 7th Ed., pg. 6-88)

$YS := 72000$ ✓ psi, yield stress of 7075-T6 Al

$$SF_u := \frac{UTS}{\sigma_t}$$

$$SF_y := \frac{YS}{\sigma_t}$$

$$SF_u = 13.14$$

$$SF_y = 11.537$$

11.0 Stress in Pivot Pin:

The pivot pin carries no load when the ball grabber is in service.

12.0 Stress in Base Plate Mounting Screws:

The load in the base plate mounting screws is due only to side loading effects because the plates are completely supported in the vertical direction by framework. In this manner, the only bolt stress is due to the 157# side load (see sketch below). The mounting screws are M10-1.5 SS capscrews with a root dia. near 0.326 in. (4 screws). The shear stress in the bolts is

$$F_h = 157 \quad \text{pounds, side force}$$

$$N := 4 \quad \text{four screws}$$

$$A_s := N \cdot \frac{\pi}{4} \cdot (0.326)^2$$

$$A_s = 0.334 \quad \text{in.}^2, \text{screw total cross-sectional area}$$

$$\tau := \frac{F_h}{A_s}$$

$$\tau = 470.235 \quad \text{psi, shear stress}$$

The safety factor is then

$$Y_S := 35000 \quad \text{psi, yield stress of 300 SS}$$

$$S_{Fy} := \frac{Y_S}{\sqrt{3} \cdot \tau}$$

$$S_{Fy} = 42.973$$

13.0 Stress in Base Plate:

The base plates are supported by either the optical support frame or the LRU frame, thereby resulting with negligible internal stresses.

14.0 Stress at Clamshell/Kinematic Ball Contact:

The load at the clamshell and kinematic ball contact line can be approximated by assuming that it is the same as the case of a cylinder in contact with a flat plate. For this case, we have (see 6th Ed. Roark, #2, pg. 651)

$$W_v := 1500 \quad \text{pounds, vertical load}$$

$$F_n := \frac{W_v}{\sin\left(30 \cdot \frac{\pi}{180}\right)}$$

$$F_n = 3 \cdot 10^3 \quad \text{pounds, surface normal load between ball and collar}$$

$$D_b := 1.969 \quad \text{in., ball diameter}$$

$$D_s := 1000000. \quad \text{in., support surface diameter (i.e., flat plate)}$$

$$c := 0.394 \quad \text{in., cutout width}$$

$$\theta := \left(\tan\left(\frac{c}{D_b} \right) \right) \cdot \left(\frac{180}{\pi} \right) \quad \text{degrees of cutout}$$

$$\theta = 21.811$$

$$CF := \frac{4 \cdot \theta}{360} \quad \text{cutout fraction} \quad CF = 0.242$$

$$LF := 1 - CF \quad \text{line contact fraction} \quad LF = 0.758$$

$$L := \left[2 \cdot \pi \cdot \left(\frac{D_b}{2} \cdot \cos\left(30 \cdot \frac{\pi}{180}\right) \right) \right] \cdot LF$$

$$L = 4.059 \quad \text{in., surface contact length}$$

$$KD := \frac{D_s \cdot D_b}{(D_s - D_b)}$$

$$KD = 1.969 \quad \text{in., contact length}$$

$$v := 0.3 \quad \text{Poisson's ratio}$$

$$E_s := 30 \cdot 10^6 \quad \text{psi, modulus of steel}$$

$$\check{E}_{\text{Al}} := 10 \cdot 10^6 \quad \text{psi, modulus of aluminum}$$

$$CE := \left(1 - v^2\right) \cdot \left(\frac{1}{E_s} + \frac{1}{E_{\text{Al}}}\right)$$

$$CE = 1.213 \cdot 10^{-7} \quad \text{per psi}$$

$$\sigma_c := 0.798 \cdot \left[\frac{\left(\frac{F_n}{L}\right)}{K_D \cdot CE} \right]^{0.5}$$

$$\sigma_c = 4.439 \cdot 10^4 \quad \checkmark \quad \text{psi, maximum compressive stress}$$

The safety factors are

$$UTS := 82000 \quad \text{psi, ultimate tensile stress of 7075-T6 Al (M. E. Handbook, 7th Ed., pg. 6-88)}$$

$$YS := 72000 \quad \text{psi, yield stress of 7075-T6 Al}$$

$$SF_u := \frac{UTS}{\sigma_c} \quad SF_y := \frac{YS}{\sigma_c}$$

$$SF_u = 1.847 \quad SF_y = 1.622$$

It should be noted that there is no safety issue associated with the low safety factors shown for the aluminum surface caused by the stainless steel ball load. Under normal load conditions, as shown above, there is a 1.62 safety factor before yielding will occur. In addition, if any condition were to occur that would cause surface yielding, the stress will be rapidly reduced to very low level as the contact diameter (i.e., surface contact area) was increased. The potential issue of fracture/crack propagation is addressed in Section 15.8 for seismic conditions.

15.0 Seismic Considerations:

The hazard category for this equipment is Category Ib, moderate or low hazard facility. This category was chosen for this equipment since failure could result with considerable onsite personnel impact, but little environmental impact. For the CAtegory Ib, the LLNL site seismic accelerations are $H = V = 0.57$. Since the equipment will be mounted to a steel and concrete structure with a low natural frequency (i.e., less than 10 Hz), the amplification factor that will be used is 2.12 at 5% damping. For these conditions, the equivalent static combined load analysis results with

$$H1 = 0.57(2.12) = 1.21$$

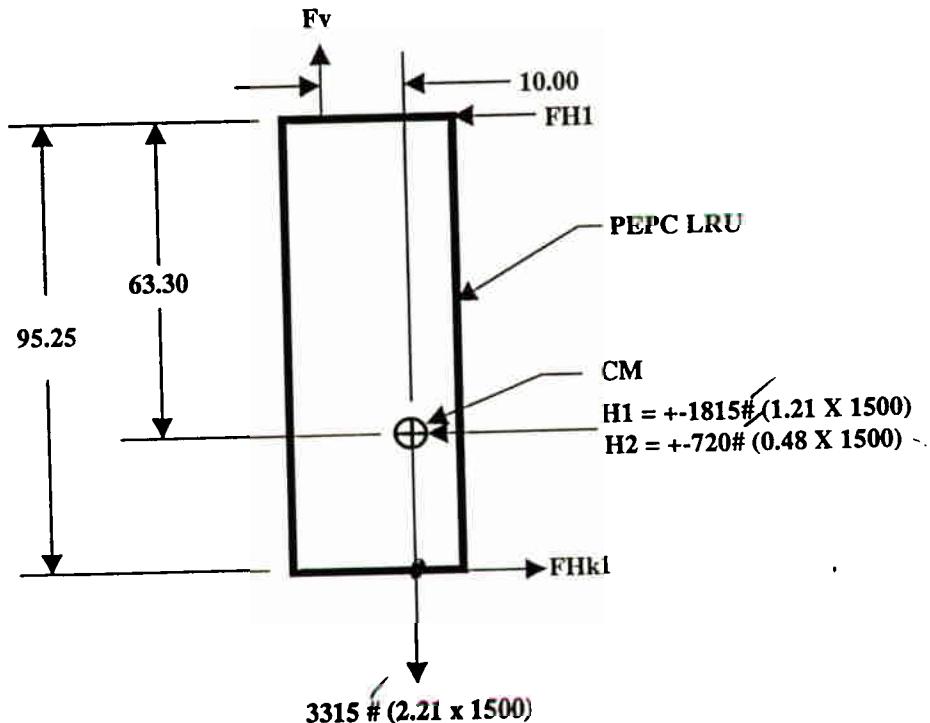
$$H2 = 0.4(H1) = 0.48$$

$$V = 0.57(2.12) = 1.21$$

Because the PEPC LRU is suspended at the top by the balls, it is free to rotate upon application of any side loads. To constrain this motion, the LRU support frame is fabricated with lower kinematic balls and seismic frame constraints that will limit transverse movement.

NOTE: ONLY THE UPPER KINEMATIC BALL SUPPORT ASSEMBLY WILL BE ANALYZED IN THIS SAFETY NOTE. THE LOWER KINEMATIC BALLS AND ASSOCIATED SEISMIC FRAME CONSTRAINTS WILL NOT BE ANALYZED HEREIN.

The increased loading due to potential seismic activity results with the load conditions



which results with the loads

$$\Sigma F_v = 0; \quad F_v = 3315 \text{ pounds (2.21X1500#)}$$

$$\Sigma F_{H1} = 0; \quad -F_{H1} + F_{Hk1} + H_{1,2} = 0$$

or $-F_{H1} = -F_{Hk1} - H_{1,2}$ (worst case of the two)

$$\Sigma M_{cm} = 0 \quad -F_v(10) + F_{H1}(63.3) + F_{Hk1}(31.95) = 0$$

or $F_{H1} = (10F_v - 31.95F_{Hk1})/63.3$

or $F_{Hk1} = (10F_v - 63.3F_{H1})/31.95$

Therefore, the worst case yields

$$-F_{H1} = -(10F_v - 63.3F_{H1})/31.95 - H_{1,2}$$

or $F_{H1} = 957 \text{ pounds } \checkmark$

Similarly, for the orthogonal axis

$$\Sigma F_{H2} = 0 \quad -F_{H2} + F_{Hk2} + H_{2,2} = 0$$

$$\Sigma M_{cm} = 0 \quad F_{H2}(63.3) + F_{Hk2}(31.95) = 0$$

or $F_{H2} = -0.505F_{Hk2}$

so $F_{H2} = F_{Hk2} + H_{2,2}$

or $F_{H2} = -1.98F_{H2} + H_{2,2}$

or $F_{H2} = 242 \text{ pounds } \checkmark$

15.1 Upper Ball Shank Seismic Stress:

As shown in Section 3.0, the stress in the upper ball shank is due to both bending and axial loads. The increase in the associated stresses due to seismic activity is linear with the increased load. Therefore, the axial seismic load stress becomes

$$F_v := 3315 \quad \text{pounds, vertical seismic load}$$

$$D := 0.78 \quad \text{inches, shank min. diameter}$$

$$\sigma_a = \frac{F_v}{\frac{\pi}{4} \cdot D^2}$$

$$\sigma_a = 6.938 \cdot 10^3 \quad \text{psi, seismic axial load stress}$$

The seismic bending stress for FH1 becomes

$$F_{H1} := 957 \quad \text{pounds, horizontal seismic load}$$

$$L := 3.85 \quad \text{inches, lever arm}$$

$$I := \frac{\pi}{4} \cdot \left(\frac{D}{2}\right)^4$$

$$I = 0.018 \quad \text{in.}^4, \text{ moment of inertia}$$

$$\sigma_b := \frac{32 \cdot F_{H1} \cdot L}{\pi \cdot D^3}$$

$$\sigma_b = 7.908 \cdot 10^4 \quad \text{psi, seismic bending stress}$$

The seismic bending stress for FH2 is less than that for the FH1, so it will not be calculated. Therefore, the maximum superposed axial stress is

$$\sigma_m := \sigma_a + \sigma_b$$

$$\sigma_m = 8.602 \cdot 10^4 \quad \text{psi, total seismic stress}$$

The safety factors are

$$\text{UTS} := 193000$$

psi, ultimate tensile stress of 17-4PH SS in H900 condition (see pg. 3.14-5, in "Damage Tolerant Design Handbook", 1983);

$$\text{YS} := 170000$$

psi, yield stress of 17-4PH SS in H900 condition

$$\text{SF} := \frac{\text{UTS}}{\sigma_m}$$

$$\text{SFy} := \frac{\text{YS}}{\sigma_m}$$

$$\text{SF} = 2.244$$

$$\text{SFy} = 1.976$$

Upper Ball Shank Fracture Critical Calculations:

$$\sigma_m = 8.602 \cdot 10^4$$

psi, tensile stress

$$K_{1c} := 85$$

ksi in², fracture toughness (see pg. 3.14-2, in "Damage Tolerant Design Handbook", 1983.)

$$a_c := \left(\frac{1}{\pi} \right) \cdot \left(\frac{K_{1c} \cdot 1000}{1.12 \cdot \sigma_m} \right)^2$$

$$a_c = 0.248$$

inches, critical flaw size

$$\frac{\sigma_m}{\text{YS}} = 0.506$$

stress ratio

For a Stress Ratio =0.08 and initial flaw size of 0.125 inches (see Figure 3.14.3.1, pg. 3.14-5, in "Damage Tolerant Design Handbook", 1983):

$$A := 2.606 \cdot 10^{-10}$$

$$n := 3.192$$

$$a_0 := 0.125 \quad \text{inches, assumed initial crack size}$$

$$N_{cycles} := \left(\frac{2}{2-n} \right) \cdot \left[\frac{1}{A \cdot \left(1.12 \cdot \frac{\sigma_m}{1000} \cdot \sqrt{\pi} \right)^n} \right] \cdot \left[ac^{\left(\frac{2-n}{2} \right)} - ao^{\left(\frac{2-n}{2} \right)} \right]$$

$N_{cycles} = 557.268$

number of seismic stress cycles required to grow
an assumed initial 0.125 inch flaw to critical size

CONDITION/HT: H900
 FORM: 0.56" TH PLATE
 SPECIMEN TYPE: CT
 ORIENTATION: L-T
 FREQUENCY: 20.00 Hz
 ENVIRONMENT: R.T., LAB AIR

YIELD STRENGTH: 170.5 KSI
 ULT. STRENGTH: 182.7 KSI
 SPECIMEN THK: 0.500"
 SPECIMEN WIDTH: 1.969"
 REFERENCES: DA001

STAIN STEEL

17-4PH

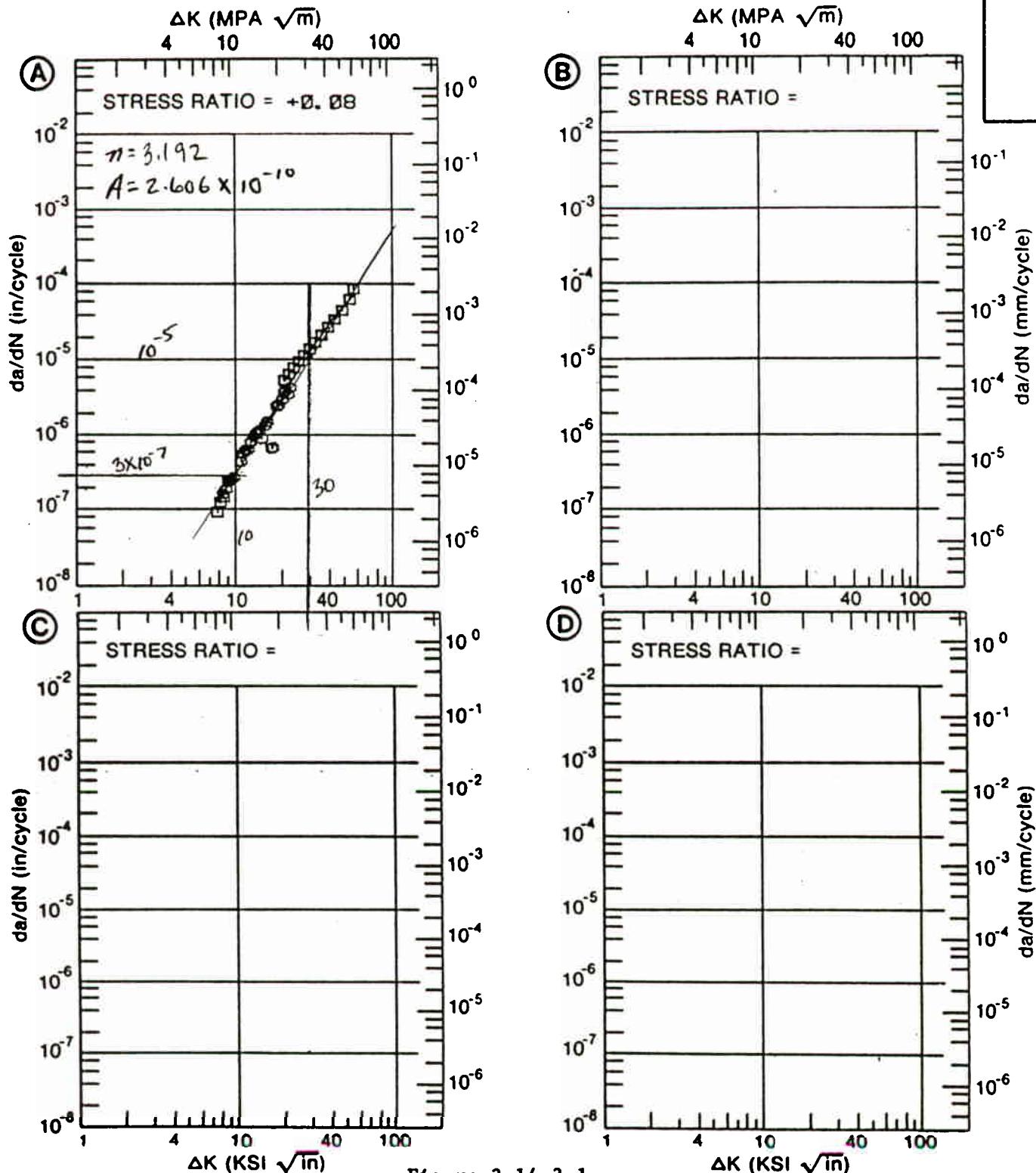


Figure 3.14.3.1

Table 3.14.2.1

CONDITION	SPECIMEN				DESIGN	CRACK LENGTH (K _{IC})/TYS ^{1/2} (IN)	K _{IC} MEAN (K _{IC} 595RT IN)	K _{IC} STAN	DATE	REFER		
	TEST FORM (IN)	THICK TEMP (F)	YIELD STRENGTH (KSI)	WIDTH THICK (IN)								
H 975	RB	3.25	R.T.	L-R	168.0	2.000	1.000	NB	1.000	0.63	84.60	----- 84212
H1025	BR	3.00	R.T.	1-L	173.3	1.990	0.503	CT	0.937	0.45	74.50	1979 DAQ01

15.2 Lower Ball Shank Seismic Stress:

The seismic stress in the lower ball shank will be less than that in the upper shank due to the reduced lever arm for load application. Therefore, it will not be determined.

15.3 Upper Translation Plate Seismic Stress:

As before (Section 5.0), the seismic stress in the upper translation plate can be calculated by superposing the stresses in a circular, fixed edge plate with central pressure loading and central moment loading. The plate seismic stress is then

$$W_v := 3315 \text{ pounds, seismic vertical load}$$

$$W_h := 957 \text{ pounds, seismic horizontal load}$$

$$L := 3.85 \text{ inches, side load moment lever arm}$$

$$M := W_h \cdot L \text{ side load moment} \quad a := 2.657 \text{ inches}$$

$$\nu := 0.3 \text{ Poisson's ratio} \quad b := 0.408 \text{ inches}$$

$$m := \frac{1}{\nu}$$

$$t := 0.75 \text{ inches, plate thickness}$$

$$\sigma_p := \frac{3 \cdot W_v \cdot (m + 1)}{2 \cdot \pi \cdot m \cdot t^2} \left[\ln\left(\frac{a}{b}\right) + \left(\frac{b^2}{4 \cdot a^2} \right) \right]$$

$$\sigma_p = 6.876 \cdot 10^3 \text{ psi, maximum plate tensile stress due to center pressure load}$$

$$k := \frac{0.1 \cdot a^2}{(b + 0.28 \cdot a)^2} \quad k = 0.532$$

$$\sigma_m := \frac{3 \cdot M}{4 \cdot \pi \cdot t^2 \cdot b} \left[1 + \left(\frac{m + 1}{m} \right) \cdot \ln \left[2 \cdot \frac{(0.45 \cdot a - b)}{0.45 \cdot k \cdot a} \right] \right]$$

$$\sigma_m = 8.351 \cdot 10^3 \text{ psi, maximum plate tensile stress due to center moment}$$

Therefore, the combined stress results with

$$\sigma := \sigma_p + \sigma_m$$

$$\sigma = 1.523 \cdot 10^4 \text{ psi, maximum plate tensile stress due to combined loads}$$

The safety factors are

UTS := 57000 psi, ultimate tensile stress of annealed 1020
 steel (LLNL M.E. DSS)

YS := 43000 psi, yield stress of 1020 steel

$$SF_u = \frac{UTS}{\sigma}$$

$$SF_y = \frac{YS}{\sigma}$$

$$SF_u = 3.743$$

$$SF_y = 2.824$$

15.4 Lower Translation Plate Seismic Stress:

The seismic stress in the lower translation plate will be less than that in the upper translation plate. Therefore, no calculations will be completed.

15.5 Tie Rod Seismic Stress:

Using the approach of Section 7.0, the seismic tie rod stress becomes

$$FH1 := 957 \text{ pounds, applied horizontal load}$$

$$M := 1.3 \cdot FH1$$

$$M = 1.244 \cdot 10^3 \text{ in. pounds, value of couple}$$

There are three tie rods, 0.375 in. in diameter, supporting the couple and effectively cantilevered. Therefore, the max. tensile stress in the rods is

$$I := \frac{\pi}{4} \cdot \left(\frac{0.375}{2} \right)^4$$

$$I = 9.707 \cdot 10^{-4} \text{ in.}^4, \text{ moment of inertia of a rod}$$

$$N := 3 \quad \text{number of rods}$$

$$\sigma := \frac{M \cdot \left(\frac{0.375}{2} \right)}{N \cdot I}$$

$$\sigma = 8.01 \cdot 10^4 \text{ psi, max. rod tensile stress}$$

This stress will obviously fail the 300 SS rods. To strengthen the rods and eliminate this failure mechanism, the rod diameter must be increased to 0.50 inches and the material changed to 1045 cold drawn steel (Again, it should be noted that a failure of the rods is not a safety issue, but one of serious inconvenience and, therefore, it must be eliminated.). The stress in the 0.50 inch dia. rods is

$$I := \frac{\pi}{4} \cdot \left(\frac{0.5}{2} \right)^4$$

$$I = 3.068 \cdot 10^{-3} \text{ in.}^4, \text{ moment of inertia of a rod}$$

$$\sigma := \frac{M \cdot \left(\frac{0.5}{2} \right)}{N \cdot I}$$

$$\sigma = 3.379 \cdot 10^4 \text{ psi, max. rod bending stress}$$

The seismic safety factors are

$UTS := 91000$ psi, ultimate tensile stress of 1045 cold drawn steel (Applied Engineering Science Handbook, 2nd Ed., page 103)

$YS := 77000$ psi, yield stress of 1045 cold drawn steel

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 2.693 \quad SF_y = 2.279$$

As before, the stress in the tie rod end screws is (see Section 7.0)

$$M = 1.244 \cdot 10^3 \text{ in. pounds, moment on a rod}$$

$$R := \frac{0.5}{2} \text{ in., rod radius}$$

$$F := \frac{M}{N \cdot R}$$

$$F = 1.659 \cdot 10^3 \text{ pounds, load per rod end screw}$$

The tensile stress in the screw (0.25-20, root dia. near 0.19 in.) is

$$\sigma := \frac{F}{\frac{\pi}{4} \cdot (0.19)^2}$$

$$\sigma = 5.851 \cdot 10^4 \text{ psi, rod end screw tensile stress}$$

The safety factor is

$UTS := 180000$ psi, ultimate tensile stress of CAMCAR allen head screws (Textron CAMCAR literature)

$YS := 162000$ psi, yield stress of CAMCAR allen head screws

$$SF_u := \frac{UTS}{\sigma} \quad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 3.077 \quad SF_y = 2.769$$

The end screw/plate thread stress is

$$L := 0.25 \quad \text{in., minimum thread engagement length}$$

$$\tau_{st} := \frac{M}{\frac{\pi \cdot 0.25 \cdot L}{N \cdot R} \cdot 2}$$

$$\tau_{st} = 1.69 \cdot 10^4 \quad \text{psi, end screw thread shear stress}$$

which results with the safety factor

$$YS := 40000 \quad \text{psi, yield stress of 6061-T6 Al (M. E. Handbook, 7th Ed., pg. 6-88).}$$

$$SF := \frac{YS}{\sqrt{3} \cdot \tau_{st}}$$

$$SF = 1.367$$

Recalculating the standard operating stress levels for the 0.50 inch dia. 1045 cold drawn steel rods and CAMCAR allen head end screws yields

$$M := 232.7$$

$$\sigma_{rod} := \frac{M \cdot \left(\frac{0.5}{2}\right)}{N \cdot I}$$

$$\sigma_{rod} = 6.321 \cdot 10^3$$

which results with the safety factors as

$$\text{UTS} := 91000 \quad \text{psi, ultimate tensile stress of 1045 cold drawn steel}$$

$$\text{YS} := 77000 \quad \text{psi, yield stress of 1045 cold drawn steel}$$

$$\text{SFu} := \frac{\text{UTS}}{\sigma_{\text{rod}}} \quad \text{SFy} := \frac{\text{YS}}{\sigma_{\text{rod}}}$$

$$\text{SFu} = 14.397 \quad \text{SFy} = 12.182$$

The end screw stress is

$$\sigma_{\text{sc}} := \frac{M}{N \cdot R \cdot \left(\frac{\pi}{4}\right) \cdot (0.19)^2}$$

$$\sigma_{\text{sc}} = 1.094 \cdot 10^4 \quad \text{psi, end screw tensile stress}$$

which results with the safety factors

$$\text{UTS} := 180000 \quad \text{psi, ultimate tensile stress of CAMCAR allen head screws}$$

$$\text{YS} := 162000 \quad \text{psi, yield stress of CAMCAR allen head screws}$$

$$\text{SFu} := \frac{\text{UTS}}{\sigma_{\text{sc}}} \quad \text{SFy} := \frac{\text{YS}}{\sigma_{\text{sc}}}$$

$$\text{SFu} = 16.449 \quad \text{SFy} = 14.804$$

The end screw/plate thread stress is

$$L := 0.25 \quad \text{in., minimum screw thread engagement length}$$

$$\tau_{\text{st}} := \frac{M}{N \cdot R \cdot \frac{\pi \cdot 0.25 \cdot L}{2}}$$

$$\tau_{st} = 3.16 \cdot 10^3$$

psi, end screw thread shear stress

which results with the safety factor

$$YS := 40000$$

psi, yield stress of 6061-T6 Al (M. E. Handbook, 7th Ed., pg. 6-88).

$$SF := \frac{YS}{\sqrt{3} \cdot \tau_{st}}$$

$$SF = 7.307$$

15.6 Pneumatic Cylinder End Screw Seismic Stress:

As in the last section and Section 8.0, the seismic stress in the pneumatic cylinder rod is

$$M := 1.3 \cdot 956$$

$M = 1.243 \cdot 10^3$ in. pounds, value of couple

$$r := \frac{0.625}{2}$$

$$I := \frac{\pi}{4} \cdot r^4$$

$$I = 7.49 \cdot 10^{-3}$$
 in.^4 , moment of inertia of the rod

$$\sigma := \frac{M \cdot r}{I}$$

$$\sigma = 5.185 \cdot 10^4$$
 psi, max. rod bending stress

The safety factors are

$$UTS := 80000 \text{ psi, ultimate tensile stress of 300SS (M. E. Handbook, 7th Ed., page 6-43)}$$

$$YS := 35000 \text{ psi, yield stress of 300 SS}$$

$$SF_u := \frac{UTS}{\sigma}$$

$$SF_y := \frac{YS}{\sigma}$$

$$SF_u = 1.543$$

$$SF_y = 0.675$$

Again, It should be noted that failure of this rod does not represent a safety issue since failure of the rod will not release the load. However, it may represent a repair issue following a seismic event.

The stress in the end screw is

$$\sigma_{sc} := \frac{1276 \cdot 1.3}{0.312 \cdot \frac{\pi}{4} \cdot (0.31)^2}$$

$$\sigma_{sc} = 7.044 \cdot 10^4 \quad \text{psi, end screw seismic tensile stress}$$

which results with the safety factors as

$$SF_u := \frac{UTS}{\sigma_{sc}} \quad SF_y := \frac{YS}{\sigma_{sc}}$$

$$SF_u = 1.136 \quad SF_y = 0.497$$

The end screw/plate thread stress is

$$L := 0.375 \quad \text{in., minimum thread engagement length}$$

$$\tau_{st} := \frac{M}{r \cdot \left(\frac{\pi \cdot 0.25 \cdot L}{2} \right)}$$

$$\tau_{st} = 2.701 \cdot 10^4 \quad \text{psi, end screw thread shear stress}$$

which results with the safety factor

$$YS := 43000 \quad \text{psi, yield stress of annealed 1020 steel (LLNL M.E. DSS)}$$

$$SF := \frac{YS}{\sqrt{3} \cdot \tau_{st}}$$

$$SF = 0.919$$

15.7 Collar Seismic Stress:

The increase in collar stress due to seismic activity is primarily due to the increased vertical load. For the increased vertical load, the horizontal component is

$$W_v := 3315 \quad \text{pounds, seismic vertical load}$$

$$F_h = \frac{W_v}{\tan\left(30 \cdot \frac{\pi}{180}\right)}$$

$$F_h = 5.742 \cdot 10^3 \quad \text{pounds, seismic horizontal force at each ball}$$

Therefore, the total horizontal force supported by the collar is

$$F_{ht} := 2 \cdot F_h$$

$$F_{ht} = 1.148 \cdot 10^4 \quad \text{pounds, total seismic horizontal force}$$

The effective pressure is then

$$H := 2.0 \quad \text{in., effective clamshell "load" height}$$

$$ID := 2.76 \quad \text{in., collar inside dia.}$$

$$OD := 3.15 \quad \text{in., collar outside dia.}$$

$$P := \frac{F_{ht}}{\pi \cdot ID \cdot H}$$

$$P = 662.194 \quad \text{psi, effective collar internal pressure}$$

The maximum collar hoop stress is then (assuming a thick walled pressure vessel)

$$\sigma := P \cdot \frac{(OD^2 + ID^2)}{(OD^2 - ID^2)}$$

$$\sigma = 5.039 \cdot 10^3 \quad \text{psi, max. collar hoop stress}$$

The safety factors are

UTS := 57000 psi, ultimate tensile stress of annealed 1020 steel (LLNL M.E. DSS)

YS := 43000 psi, yield stress of 1020 steel

$$SF_u := \frac{UTS}{\sigma} \qquad SF_y := \frac{YS}{\sigma}$$

$$SF_u = 11.311 \quad SF_y = 8.533$$

15.8 Clamshell Seismic Stress:

The seismic stress in the clamshell can be determined by treating it as a beam with an applied axial force (i.e., the 3315 # seismic vertical load) that creates a tensile stress due to both an axial load and an end couple as was done in Section 10.0. As before, the predicted stresses are conservative.

The uniform stress caused by the axial load can be calculated from

$$W_v := 3315 \quad \text{pounds, vertical load}$$

$$ID := 2.166 \quad \text{in., inside diameter} \quad R_i := \frac{ID}{2} \quad R_i = 1.083$$

$$OD := 2.756 \quad \text{in., outside diameter} \quad R_o := \frac{OD}{2} \quad R_o = 1.378$$

$$c = 0.394 \quad \text{in., cutout width}$$

$$\theta := \left(\tan^{-1} \left(\frac{c}{\frac{OD + ID}{4}} \right) \right) \cdot \left(\frac{180}{\pi} \right) \text{ degrees of cutout}$$

$$\theta = 17.755$$

$$CF := \frac{4 \cdot \theta}{360} \quad \text{cutout fraction} \quad CF = 0.197$$

$$AF := 1 - CF \quad \text{area fraction} \quad AF = 0.803$$

$$\sigma_a := \frac{W_v}{\frac{\pi}{4} \cdot (OD^2 - ID^2) \cdot AF}$$

$$\sigma_a = 1.811 \cdot 10^3 \quad \text{psi, axial stress from axial load}$$

The maximum tensile stress due to the end couple is

$$\theta_{max} := \left(\tan^{-1} \left(\frac{c}{\frac{ID}{2}} \right) \right) \cdot \left(\frac{180}{\pi} \right) \quad \text{degrees, cutout angle at minimum radius}$$

$$\theta_{max} = 19.992$$

$$\theta_{max} := \theta_{max} \cdot \frac{\pi}{180} \quad \text{radians} \quad Y_o := c \quad \text{in., cutout width}$$

$$A := \left(\int_{Ri}^{Ro} r dr \right) \cdot \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$A = 0.887$ in², approx. area of minimum thickness section

$$YDA := \left[\int_{\theta_{max}}^{\pi - \theta_{max}} \sin(\theta) d\theta \right] \cdot \left(\int_{Ri}^{Ro} r^2 dr \right) - Yo \cdot \left(\int_{Ri}^{Ro} r dr \right) \cdot \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$YDA = 0.494$ in³, integral of $y^2 dA$

$$Ycm := \frac{YDA}{A}$$

$Ycm = 0.557$ in., distance to center of mass (area) from the cutout

$Y := Ycm + Yo$ $Y = 0.951$ in., distance to center of mass from clamshell center

$$Int1 := Y^2 \cdot \left(\int_{Ri}^{Ro} r dr \right) \cdot \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (1) d\theta \right]$$

$$Int2 := 2 \cdot Y \cdot \left(\int_{Ri}^{Ro} r^2 dr \right) \cdot \left[\int_{\theta_{max}}^{\pi - \theta_{max}} \sin(\theta) d\theta \right]$$

$$Int3 := \left(\int_{Ri}^{Ro} r^3 dr \right) \cdot \left[\int_{\theta_{max}}^{\pi - \theta_{max}} (\sin(\theta))^2 d\theta \right]$$

$Ixx := Int1 - Int2 + Int3$

$Ixx = 0.058$ in.⁴, beam bending moment of inertia, i.e., integral of $y^2 dA$

$$L := \frac{0.295}{2} + \left(1.083 - \frac{1.969}{2} \cdot \cos\left(30 \cdot \frac{\pi}{180}\right) \right)$$

$L = 0.378$ in., couple force lever arm to clamshell wall centerline

$$M := Wv \cdot L$$

$$M = 1.253 \cdot 10^3 \text{ in. pounds, applied end couple}$$

$$\sigma_m := \frac{M \cdot Y_{cm}}{I_{xx}}$$

$$\sigma_m = 1.198 \cdot 10^4 \text{ psi, max. tensile stress due to end couple}$$

Therefore, the total max. stress is the superposition of the above two stresses, i.e.,

$$\sigma_t := \sigma_a + \sigma_m$$

$$\sigma_t = 1.379 \cdot 10^4 \text{ psi, total superposed tensile stress}$$

The safety factors are

$UTS := 82000$	psi, ultimate tensile stress of 7075-T6 Al (M. E. Handbook, 7th Ed., pg. 6-88)
$YS := 72000$	psi, yield stress of 7075-T6 Al

$$SF_u := \frac{UTS}{\sigma_t} \qquad SF_y := \frac{YS}{\sigma_t}$$

$$SF_u = 5.946 \qquad SF_y = 5.221$$

Clamshell Fracture Critical Calculations:

$$\sigma_t = 1.379 \cdot 10^4 \text{ psi, tensile stress at minimum wall thickness}$$

$$K_{1c} := 24.5 \text{ ksi in}^2, \text{fracture toughness (see LLNL ME DSS, Section 5, Table 1.)}$$

$$ac := \left(\frac{1}{\pi} \right) \cdot \left(\frac{K_{1c} \cdot 1000}{1.12 \cdot \sigma_t} \right)^2$$

$a_c = 0.801$ inches, critical flaw size

$\frac{\sigma_t}{Y_S} = 0.192$ stress ratio

For a Stress Ratio = -1.00 and initial flaw size of 0.125 inches (see Figure 8.9.3.6A, pg. 8.9-89, in "Damage Tolerant Design Handbook", V.2, 1983):

$$A = 5.29 \cdot 10^{-12}$$

$$n = 5.679$$

$a_0 = 0.125$ inches, assumed initial crack size

inches, critical flaw size

$$N_{cycles} := \left(\frac{2}{2-n} \right) \cdot \left[\frac{1}{A \cdot \left(1.12 \cdot \frac{\sigma_t}{1000} \cdot \sqrt{\pi} \right)^n} \right] \cdot \left[a_c \left(\frac{2-n}{2} \right) - a_0 \left(\frac{2-n}{2} \right) \right]$$

$N_{cycles} = 3.13 \cdot 10^4$ number of seismic stress load cycles required to grow and assumed 0.125 inch flaw to critical size for a stress ratio of -1.0

Similarly, for a Stress Ratio = 0.5 and initial flaw size of 0.125 inches (see Figure 8.9.3.6C, pg. 8.9-89, in "Damage Tolerant Design Handbook", V.2, 1983):

$$A := 1.43 \cdot 10^{-8}$$

$$n := 3.23$$

$a_0 = 0.125$ inches, assumed initial crack size

inches, critical flaw size

$$N_{cycles} := \left(\frac{2}{2-n} \right) \cdot \left[\frac{1}{A \cdot \left(1.12 \cdot \frac{\sigma_t}{1000} \cdot \sqrt{\pi} \right)^n} \right] \cdot \left[a_c \left(\frac{2-n}{2} \right) - a_0 \left(\frac{2-n}{2} \right) \right]$$

$N_{cycles} = 6.33 \cdot 10^3$ number of seismic stress load cycles required to grow and assumed 0.125 inch flaw to critical size for a stress ratio of 0.5

CONDITION/HT: 116
 FORM: ~~THIN PLATE~~
 SPECIMEN TYPE: CCP
 ORIENTATION: L-T
 FREQUENCY: 10.00 Hz
 ENVIRONMENT: R.T., L.H.A.

YIELD STRENGTH: 80.0 KSI
 ULT. STRENGTH: 88.0 KSI
 SPECIMEN THK: 0.250"
 SPECIMEN WIDTH: 4.000"
 REFERENCES: MA007

ALUM.
ALLOY

7075

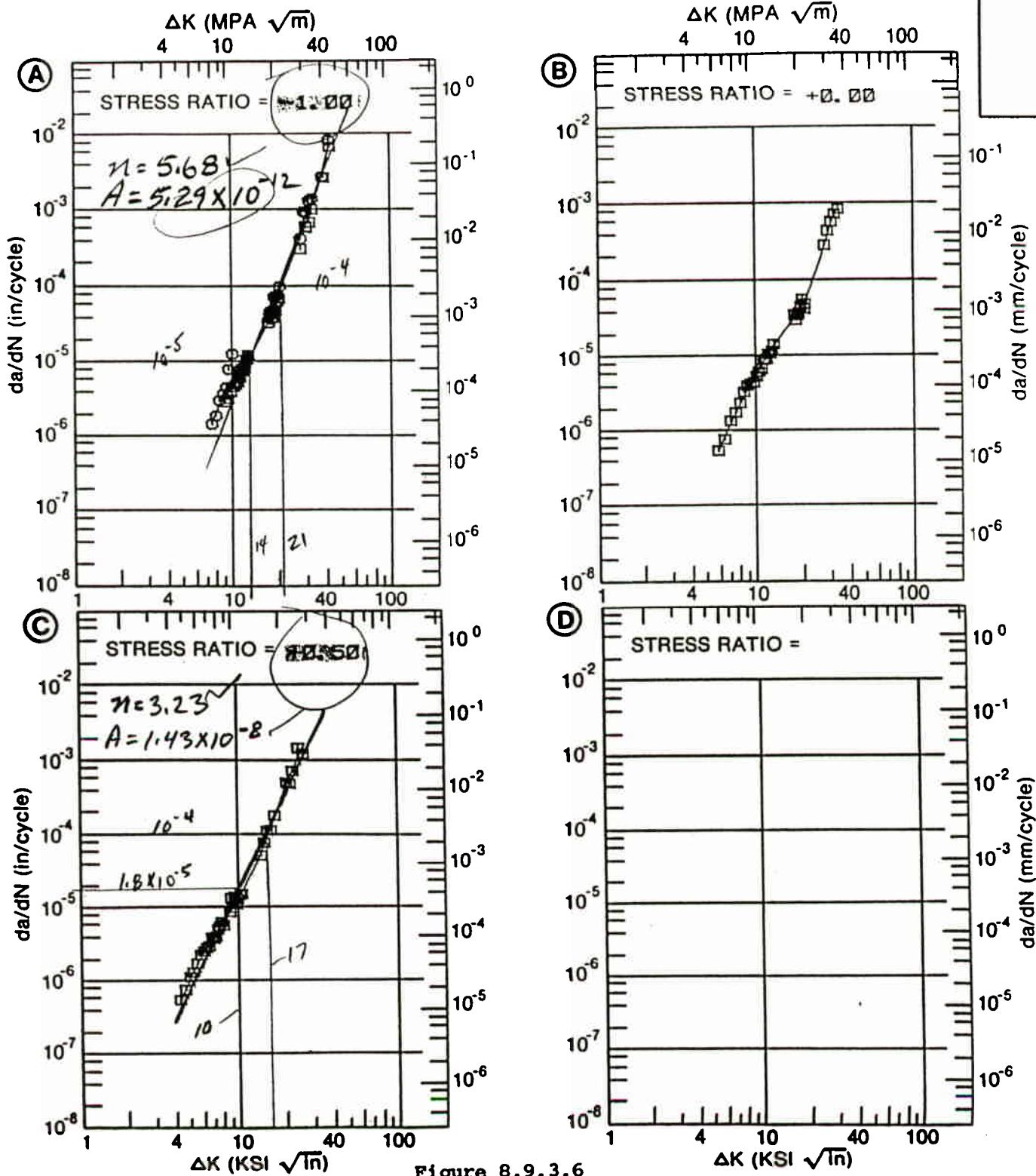


Figure 8.9.3.6

TABLE KAR1. (CONTINUED)

ALLOY	CONDITION	FORM AND THICK IN.	TEST TEMP F	AVERAGE YIELD STR SQ KSI	SPECIMEN IDENT	SPEC DESIGN	SPECIMEN DIMENSIONS			KIC	2.5
							(B)	THICK (W)	WIDTH (IN.)	KSI-SQ IN.	(KIC/YI)SQ IN.
				82 66.0	411051	CT	L-T	2.319	4.999	2.757	43.6
				82 66.0	411051	CT	L-T	2.021	4.061	2.122	40.2
				82 66.0	411051	CT	L-T	2.019	4.000	2.120	41.2
										AVG 40.7	
SUMMARY							L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
<u>7075 1651</u>	P	70-89					T-L	<u>26.45</u> <u>1.70</u> <u>29</u>			
7075	T7351	P	70-89				L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							T-L	21.44 1.44 36			
7075	T7651	(SP)	70-89				L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							T-L	29.14 1.98 23			
							T-L	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							L-T	29.52 6.96 25			
							L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							T-L	27.05 1.49 41			
							L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							T-L	23.61 1.62 41			
							L-T	OVERALL AVERAGE KIC STANDARD DEVIATION NUMBER OF SPECIMENS			
							T-L	40.63 2.75 10			

(SP)=SPECIAL PROCESSING

15.9 Pivot Pin Seismic Stress:

The pivot pin carries no load, therefore, no calculations will be completed.

15.10 Base Plate Bolts Seismic Stress:

The seismic load in the mounting screws is due only to the side loading since the plates are mounted to the back side of the support frames. In this manner, the only bolt stress from the applied loads is due to the 957# side load. According to the methods of Section 12.0, the shear stress in the bolts is

$$F_h := 957 \quad \text{pounds, side force}$$

$$N := 4 \quad \text{four screws}$$

$$A_s := N \cdot \frac{\pi}{4} \cdot (0.326)^2$$

$$A_s = 0.334 \quad \text{in.}^2, \text{screw total cross-sectional area}$$

$$\tau := \frac{F_h}{A_s}$$

$$\tau = 2.866 \cdot 10^3 \quad \text{psi, shear stress}$$

The safety factor is

$$Y_S := 35000 \quad \text{psi, yield stress of 300 SS}$$

$$S_{Fy} := \frac{Y_S}{\sqrt{3} \cdot \tau}$$

$$S_{Fy} = 7.05$$

15.11 Base Plate Seismic Stress:

The seismic stress in the base plate is negligible since it is fully supported by frame members.

15.12 Seismic Stress at Clamshell to Kinematic Ball Contact:

The seismic loading at the clamshell to kinematic ball contact line can be determined according to the methods of Section 14.0 as

$$F_v := 3315 \quad \text{pounds, seismic vertical load}$$

$$F_n := \frac{F_v}{\sin\left(30 \cdot \frac{\pi}{180}\right)}$$

$$F_n = 6.63 \cdot 10^3 \quad \text{pounds, seismic surface normal load}$$

$$D_b := 1.969 \quad \text{in., ball diameter}$$

$$D_s := 1000000. \quad \text{in., support surface diameter}$$

$$c := 0.394 \quad \text{in., cutout width}$$

$$\theta := \left(\operatorname{atan} \left(\frac{c}{D_b} \right) \right) \cdot \left(\frac{180}{\pi} \right) \quad \text{degrees of cutout}$$

$$\theta = 21.811$$

$$CF := \frac{4 \cdot \theta}{360} \quad \text{cutout fraction} \quad CF = 0.242$$

$$LF := 1 - CF \quad \text{line contact fraction} \quad LF = 0.758$$

$$L := \left[2 \cdot \pi \cdot \left(\frac{D_b}{2} \cdot \cos\left(30 \cdot \frac{\pi}{180}\right) \right) \right] \cdot LF$$

$$L = 4.059 \quad \text{in., surface contact length}$$

$$KD := \frac{D_s \cdot D_b}{(D_s - D_b)}$$

$$KD = 1.969$$

$$v := 0.3 \quad \text{Poisson's ratio}$$

$$E_s := 30 \cdot 10^6 \quad \text{psi, modulus of steel}$$

$$E_{Al} := 10 \cdot 10^6 \quad \text{psi, modulus of aluminum}$$

$$CE = \left(1 - v^2\right) \cdot \left(\frac{1}{E_s} + \frac{1}{E_{Al}}\right)$$

$$CE = 1.213 \cdot 10^{-7}$$

$$\sigma_c := 0.798 \cdot \left[\frac{\left(\frac{F_n}{L} \right)}{K_D \cdot CE} \right]^{0.5}$$

$$\sigma_c = 6.599 \cdot 10^4 \quad \text{psi, maximum compressive stress}$$

The safety factors are

$$UTS := 82000 \quad \text{psi, ultimate tensile strength of 7075-T6 Al}$$

$$YS := 72000 \quad \text{psi, yield strength of 7075-T6 Al}$$

$$SF_u := \frac{UTS}{\sigma_c} \qquad SF_y := \frac{YS}{\sigma_c}$$

$$SF_u = 1.243 \qquad SF_y = 1.091$$